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Thermal Interaction in a Refrigeration Twin Screw Compressor During the Compression Process.

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Abstract

In this paper a mathematical model for the process of heat interaction between the compressed gas and the screws in a twin screw compressor is presented and discussed. The simulation process is based on the results of the working process simulation which was developed previously at the University of Strathclyde. The simultaneous influence of geometrical and physical characteristics of the screws, the working process and the thermophysical properties of the gas are considered. The influence of different parameters on the thermal interaction is discussed. Rotor thermal distortion analysis is provided.

Introduction

Around the world many engineers are working to develop and refine the twin screw compressor with a view to maintaining or increasing its competitive advantage over other types. Other engineers aim at more general research objectives, for example to develop useful analytical techniques and to increase understanding of the behavior of the machine generally. This paper is presented as a contribution principally to the latter objective but has a practical application in helping to identify deflections caused by different physical mechanisms; e.g. heat and gas pressure.

The surfaces enclosing the compression cavity are the housing bores, the end plates and the helical surfaces of the rotors (see Fig.1). The heat exchange between cavity gas and these surfaces is complex because not only do the surface areas change, the actual area locations change; i.e. they travel and of course reduce in area to effect the compression.

For the simulation, the following assumptions and simplifications were made:

1. During the compression process the gas exchanges heat with the housing and screw surfaces. In this model only heat exchange between gas and screws is considered.

2. Heat fluxes from/to the rotor along the shafts at the suction and discharge ends were neglected, as were the heat interchange processes between the rotors and its bearings and seals. They were considered to be small compared with heat exchange with the gas.

3. The rotor was discretized into N_s cross sectional slices, perpendicular to the main axis. In each slice two rotor zones were introduced each having a uniform temperature: the circular rotor central piece and the lobe.

4. During compression, the gas temperature rises much faster than that of the rotors, thus the effect on the compression process of the temperature difference between two neighbouring lobes is negligible.

5. The thermodynamic properties of the gas in the cavity were taken to be uniform throughout the cavity volume.

Mathematical Modelling

The rotor discretization facilitated the formation of the heat exchange matrix A and the heat load vector B. Using rotor geometrical profile data provided by computer programs [1], numerical procedures give the volume, outer surface, distance between centers of gravity of the domains and effective contact area. The discretized heat conduction model is represented by matrix A and heat load vector B, as follows:

$$\frac{d}{dt} T = A \times T + B \times q$$

Where T is the vector of temperatures, $2N_s$ in size

q is the heat flux on the outer surface of the domain. $q_i = h \times (T_g - T_i)$

where h is the heat transfer coefficient and T_g the gas temperature.

The working process temperatures were taken from the thermodynamic simulation program described in [2].

According to [3], a reasonable estimate of the heat transfer coefficient between the gas and rotor can be determined from the simple equation: $Nu=4.36$. For R-22 refrigerant and the screw geometry, the following heat exchange coefficient was calculated: $h_0 = \frac{22.36 W}{m^2 deg C}$

During the compression process the density of the gas changes considerably. To account for this the local heat transfer coefficient was calculated as follows: $h = h_0 \times V_0 / V$

where V_0 is the specific volume of the gas at suction conditions.

The surface of the cavity volume changes during the compression process, in such a manner that the i -th cross section has a contact with the compressing gas, until $\varphi_i < 2\pi$.

where $\varphi_i = \phi + W_a \times (i - 0.5) / N_s$

ϕ is the rotation angle, rad;

W_a is the rotor wrap angle, rad;

i is the cross-section number.

In the case of $\varphi_i \geq 2\pi$ the rotor domain was exchanging heat with the gas at the suction conditions.

Heat exchange between the male and female rotors was assumed to take place in the cross section of the male/female rotor, for which $2\pi - F_k \leq \varphi_i \leq 2\pi$, where F_k =heat interchange angle, which was determined by the rotor profile analysis.

The values of heat fluxes between each of the cross sections and the gas during the rotation process were used to calculate the total heat flux between the cavity volume considered and the rotor. The contact between a rotor and the compressing gas takes place from $\phi = -W_a$ up to $\phi = 2\pi$.

This heat flux can be calculated according to the formula:

$$H(\phi) = \sum_{i=i_2}^{i_1} Q(\phi)$$

where i_1 and i_2 depend on ϕ in the following way:

In the case of $-W_a < \phi \leq 0 \Rightarrow i_1 = N_s; i_2 = INT(0.5 - N \times \phi / W_a)$

In the case of $0 < \phi \leq 2\pi - W_a \Rightarrow i_1 = N_s; i_2 = 1$

In the case of $2\pi - W_a < \phi \leq 2\pi \Rightarrow i_1 = INT(N_s \times (2\pi - \phi) / W_a + 0.5); i_2 = 1$

Simulation process

The simulation was achieved via a group of computer programs written for an IBM 386 PC computer using Turbo Pascal language. All of them use the same data files and can be started using the general menu program.

The programs provide the following:

- 1.The formation of the A and B matrices using the rotor profile data for the male and female rotors.
- 2.Transient thermal analysis for male and female rotors separately.
- 3.Transient thermal analysis for male and female rotors simultaneously.

4.Heat exchange between the gas and the rotors (for a cavity volume, as a function of male rotor rotation angle).

5.Thermal distortion of male and female rotors.

Computer programs 2 and 3 are based on the simplifications already described. The equation (1) was integrated, using the Runge-Kutt (IV) method. The initial conditions (for the time zero) was that the temperature of the rotor was equal to the suction temperature. There were 90 time steps per one male rotor revolution. The integration was continued until: 1)The rotor temperature at the end of the discharge process becomes constant; 2) Heat balance between rotor and gas becomes zero.

The input data was the following:

- male and female rotor profile data;
- temperature and specific volume diagrams vs. male rotor rotation angle;
- wrap angle and rotor length, male:female rotor lobe ratio;
- compressed gas characteristics;
- rotor material properties (density, thermal conductivity and capacity, thermal linear expansion coefficient).

Results of the simulation

On Fig.2 the final temperature distribution is presented both for the male and female rotors. A small difference between the lobe and central temperature in each cross section is predicted and seems reasonable for this simplified approach. At the same time one can see, that the temperatures in the female rotor are less for any chosen slice than those in the male one. This occurs because the wrap angle of the female rotor is $2/3$ of that of the male wrap, and the compression time, i.e. time of rotor heating, for one revolution, is less than that of the male rotor.

The calculations show that the effect of rotor-rotor heat transfer due to physical contact at the contact line is very small indeed and may be neglected.

Since the temperature field in a rotor is considered to be one dimensional, (the temperature change within each cross section is very small), thermal distortion of the rotor can be calculated very easily in each of the cross sections using the linear expansion model. For steel rotors the radial linear expansion at the hot end is of magnitude of 0.01 mm, which is of the same order as rotor-rotor and rotor-housing clearance. The radial and axial distortions of the male and female rotors are presented in Fig.3 and Fig.4 respectively.

To prove the adiabatic assumption of the thermodynamic process model, a comparison between gas-rotor heating and an increment of internal gas energy was made. From Fig.5 one can see that gas heating to and from the rotor is a relatively small fraction of total internal energy change caused by compression of the gas in the cavity.

Another parameter, which is being determined with a relatively large error, is the heat transfer coefficient between gas and rotor. Several calculations were done covering a wide range of heat transfer coefficient values, from 0.5 up to 1.5. Fig.6 represents the dependance of the male rotor hot end temperatures on the heat transfer coefficients used in the model.

Summary

1.The effect of heat interchange between rotor and gas is not large compared with the cavity gas internal energy, and it need not be taken into account in thermodynamic models of compressor behavior.

2.In a refrigeration compressor the temperature rises of the rotors are relatively small, but nevertheless give rise to deformations which are comparable with the clearances and consequently are capable of affecting the leakage considerably. In dry machines where the temperature rises are greater, this effect could be greater. Further investigation of the effect on clearances on thermal distortions is recommended.

Acknowledgment

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References

- 1.Yan Tang and John S.Fleming,Obtaining the Optimum Geometrical Parameters of a Refrigeration Helical Screw Compressor. International Compressor Engineering Conference at Purdue, 1992.
- 2.Yan Tang and John S.Fleming, Simulation of the Working Process of the Oil Flooded Helical Screw Compressor with Liquid Refrigerant Injection.International Compressor Engineering Conference at Purdue, 1992.
- 3.Shane Harte,Thermodynamic Design Analysis of Refrigeration Compressors. Ph.D. Thesis, University of Dublin, Trinity College, Dublin, Republic of Ireland, 1993.

Fig.1.Graphic representing compression cavities of the rotors.

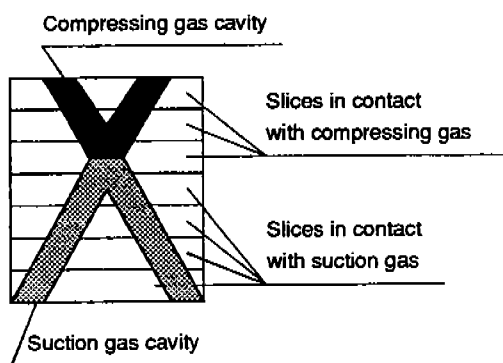


Fig.2.Temperature in male and female rotors vs. element location (deg C).

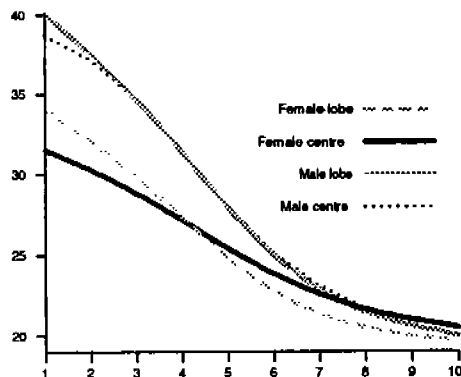


Fig.3.Thermal distortion in male rotor (mm) vs. element location.

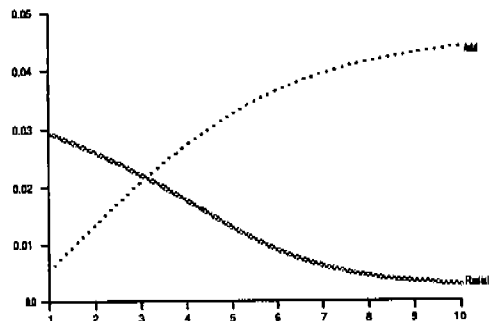


Fig.4.Thermal distortion of female rotor (mm) vs. element location.

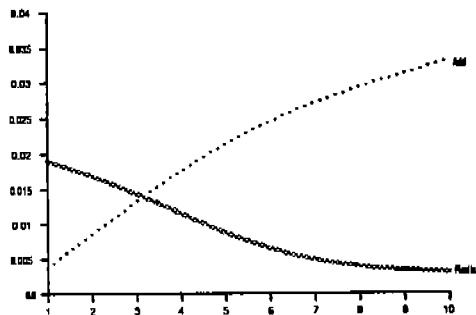


Fig.5.Rotor-gas heating compared with gas specific energy increment vs. male rotor angle (deg).

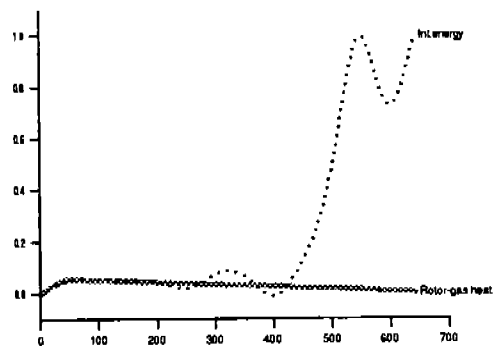


Fig.6.Rotor hot end temperature (C) vs.heat exchange coefficient variation.

